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Barry Hudson

AUSTRALIA

Patents Act 1990

PROVISIONAL SPECIFICATION

for the invention entitled:

"A rotary engine"

The invention is described in the following statement:

A ROTARY ENGINE

This invention relates to a rotary engine, and more particularly to a rotary engine that produces power from pure rotary motion.

Through the history of reciprocating engines, immense effort has been expended in an effort to reduce the effect of vibration. In trying to overcome this vibration, the careful design of the counterweighing of a crankshaft is vital, as is matching the static weights of the pistons, gudgeon pins and conrods. Dynamic balancing of the crankshaft assembly is fundamental to this exercise since its action is essentially eccentric. Potentially damaging critical frequency vibration zones, amid the revolution range, are usually changed by adding a harmonic balancer, of a selected mass, to the front of the crankshaft.

Mitsubishi for example, has developed an additional chain driven balance shaft to help negate the deleterious sensation of vibration. It is very effective, but it is after the act, and justifiably, absorbs an additional amount of power. It would be much better if a device of this type was not necessary.

In addition to the low frequency vibration associated with reciprocating engines, there is also a substantial amount of higher frequency audible noise generated by the valve drive train. The timing chain, camshaft/s, cam-followers, rockers, tappets, valves and valve springs all contribute to the noise level emitted.

Reciprocating engines also have a crankshaft with a conrod, gudgeon pin, and piston assembly, which moves through a nearly sinusoidal acceleration – deceleration cycle, from momentarily stationary at the top, to maximum speed in the middle, to stationary at the bottom, to maximum speed again at the middle, to stationary again at the top. One of the major considerations when designing a reciprocating engine is the amount of conrod flex, and this by itself indicates how much energy is needlessly expended.

Rotary gear engines having a male rotor with lobes (also referred to herein as teeth) co-operating with a female rotor having cavities, produce power from a relatively pure rotary motion and largely remove the vibration alluded to above. However, previously proposed engines of this type present the additional problem of adequately removing the gases burnt in the combustion phase from the chamber during the exhaust phase.

In accordance with the present invention, there is provided a rotary engine comprising a housing having a male rotor having a plurality of projecting lobes and a female rotor having a plurality of cavities, the male and female rotors being mounted for synchronous rotation about parallel axes such that during rotation successive lobes on the male rotor mate with successive cavities on the female rotor to define therewith a combustion chamber in which a mixture of air and fuel is compressed by the interaction of the lobe and the cavity during rotor rotation, at least one exhaust port leading out of the housing for discharge of exhaust gases from the cavity of the female rotor following combustion and from the space between adjacent lobes of the male rotor following combustion, and respective purge ports leading out of the housing downstream of the exhaust port in the direction of rotor rotation to facilitate discharge of residual exhaust gases from the cavity and inter-lobe space , the purge ports being associated with air inlet ports to admit air into the cavity and inter-lobe space in preparation for the subsequent combustion cycle.

According to the preferred embodiment of the present invention there is a separate exhaust port for the male and female rotor. Advantageously, discharge of the residual exhaust gases via the purge ports occurs under the effect of centrifugal force generated by rotor rotation; for this purpose the purge ports lead radially out of the housing.

Preferably the purge ports extend over a relatively large arc of the order of 90° to 120°. In the preferred embodiment intake of air during purging occurs via intake ports in at least one end wall, and preferably both end walls, of the rotor housing.

Embodiments of the present invention will now be described, by way of example only.

To facilitate a better understanding of embodiments according to the invention, a brief background to the development is included.

Conception

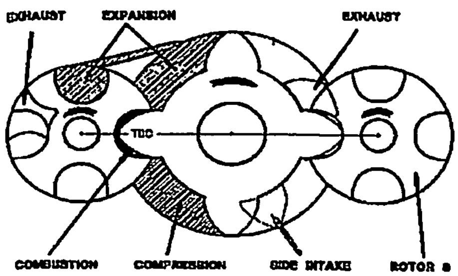
All engines and compressors need a differential area between an active and reactive element. The ideal would be a device which had a single rotor on a shaft and was able to induct a fresh air/fuel charge, compress it into a combustion chamber, then utilize the expansion against a reactive element to produce torque, then exhaust the burnt gas. All this should happen in one revolution.

The main problem was to devise a suitable reactive element.

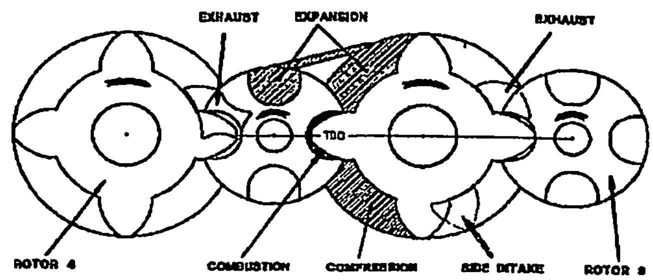
The original profiles were intended to be extended in a helical form, so that the compressed gas is forced axially along the helical screw and into an exhaust port at the far end. However, if they were straight cut, they would satisfy the present requirement. The charge could be compressed between the male tooth and a slightly enlarged female profile. The amount of this enlargement would determine the compression ratio.

The initial manifestation concept was to use a normal 4 stroke Otto cycle which fires every revolution. It would be completely rotary, with no reciprocating parts. The first design had three sets of shafts and rotors. The third shaft and rotor was to purge the burnt gas from the chambers in the male rotor and induce a fresh charge. A fourth shaft and rotor would be required to purge the chambers in the female rotor. This would be very unwieldy.

The diagram below shows the initial concept using a ratio of 1:1 with 4 teeth.



3 ROTOR



4 ROTOR

In the three rotor diagram, the left-hand rotor is the active female rotor, and after exhaust, a full chamber of burnt gas remains. To purge this gas a forth rotor would be necessary. From the four rotor diagram, it was obvious that there was excess rotation doing nothing. If this space had an additional port, which might allow this volume of gas to be centrifugally discharged, and if there were a two further a side ports, open to atmosphere, in the endplates, close to the root diameter, fresh air might be drawn in to replace the burnt gas. Still further, if this would work for the third and fourth rotors, it might also work for the two main rotors. This would eliminate the need for the third or fourth rotors. This constitutes an additional phase.

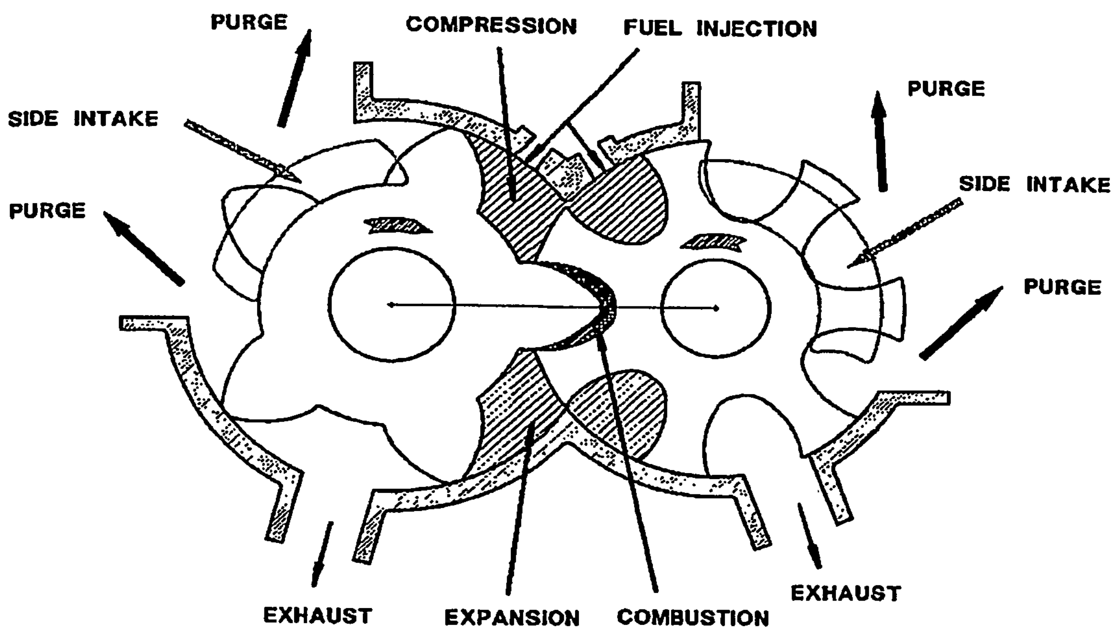
A five cycle engine - The fifth phase would require more chambers to allow all five cycles to be enacted in one revolution, however, having a larger number of chambers would also produce a smoother torque output.

From the problems and reasoning discussed above, embodiments according to the present invention were developed.

Principle.

Two rotors are fixed to two shafts with a pair of timing gears to maintain synchronization and prevent clash. The rotors each have a particular profile and interact with each other in a similar fashion to a pair of spur gears. The rotors are enclosed in a binocular shaped main housing with two endplates. Two bearings in each endplate carry the shafts. The main

housing has two primary and two secondary exhaust ports. The inlet ports are located in the endplates. Two spark plugs are also located in the endplates.



As the rotors rotate, air is trapped in the spaces between the teeth and is compressed as the rotors progress. Fuel is injected during this compression cycle. Maximum compression is reached when the tooth of male rotor is aligned with the gap in the female rotor. A spark plug is located at each end of the compression cavity and fire simultaneously at the appropriate time. The expansion of the gas raises the pressure and forces the rotors to rotate. The cavity formed by the rotors expands as rotation progresses until maximum volume is reached. At this stage, the cavities in each rotor reach a primary exhaust port in the main housing. The residual pressure in the cavities is released as the cavities pass across the exhaust ports. However, the cavities still contain burnt gas. As rotation progresses, the cavities in both rotors each reach a secondary exhaust port in the main housing. These secondary exhaust ports are quite long and large. The residual burnt gas in the cavities is centrifugally thrust into these ports and the cavities then reach ports in the endplates, which admit clean air. The centrifugal action of the burnt gas moving radially outward, draws clean

air into the cavities. The cavities then pass the end of the secondary ports, the clean air is confined by the main housing and the compression cycle recommences.

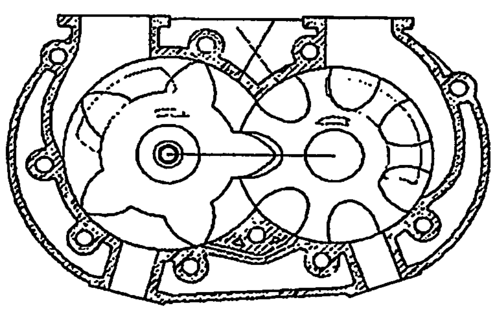
All five cycles are happening simultaneously, and each cavity in the male rotor fires every revolution. Two expansion and two compression cycles occur concurrently for 30° of rotation for each cavity.

Preliminary Design 1000cc

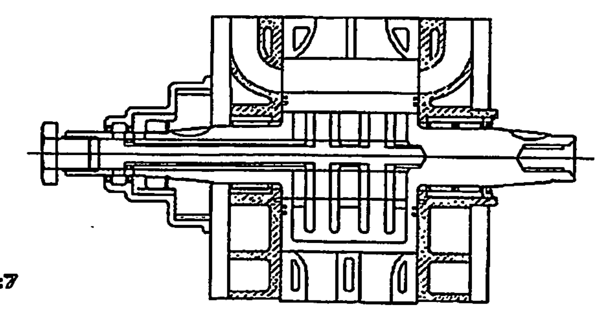
Profile Refinement.
The profile could be illustrated as the path described by the locus of one rotor as it intrudes on the other. The larger rotor assumes a male, or convex, profile while the smaller adopts a female, or concave, contour. A pair of profiles is acceptable when there is no overlap or undercut in the male. There are many acceptable ratios. The ratio, number of teeth and maximum tooth depth are interdependent factors.

Ratio Investigation.
The next exercise was to explore the factors controlling the various gear ratios, and number of teeth which might prove practical for this application. The larger rotor is always the male. A larger number of teeth are required to fit all five cycles into 360°, while a lesser number of teeth, allows a greater tooth depth and simplicity.
The most promising ratio was shown to be 5:7.

Part of the preliminary design of an engine of 1000cc capacity at 5:7 ratio.



5:7

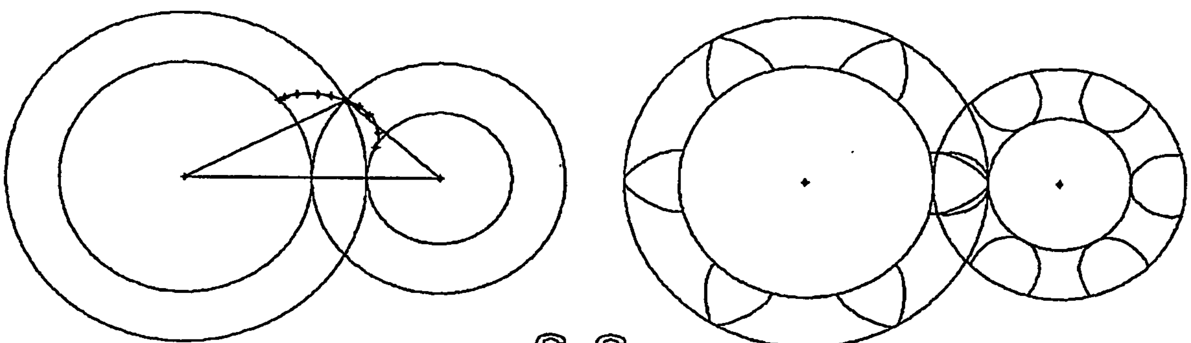


The 1:1 Ratio.

The main factor involved is the width of the female teeth at their narrowest point. This is governed by the number of teeth in the female rotor, relative to diameter and tooth depth. Therefore, a larger diameter female, proportionate to the male, seemed to be indicated. The 2:1 ratio allows equal diameter rotors. As the ratio approaches 1:1, the female diameter reduces and the male diameter increases. The female tooth width is governed by the tooth depth. A larger number of teeth demands a smaller tooth depth for a given diameter.

In proceeding, the female, with a reasonable tooth width, was used to determine the factors governing the proportions of 1:1. The male diameter was quite large , but six teeth did seem to be practicable. If there must be two rotors, it would be better if one of them had as little mass as possible. A larger male root diameter would allow freedom of choice of the main shaft size. A reasonable tooth depth and width was practicable.

The following diagram shows he preferred proportions.



6:6

1:1 looked to be practical with six teeth in both the male and female rotors. Six teeth would allow a longer purge cycle. The female tooth width appeared satisfactory and the root diameter is large enough to permit suitable shaft size and end sealing.

Preferably, at least five teeth are provided for a normally aspirated engine. If a turbo-charger or supercharger were to be fitted, at least seven or eight teeth would be preferable, as the fifth

cycle would need to be shortened to prevent the pre-compressed air entering during the purge stage. The addition of two intake ports, just prior to the commencement of the compression stage, would allow separate admission of the pre-compressed air. An additional attribute is that exhaust back-pressure does not effect the torque output of the engine. This makes it ideal for turbo-charging.

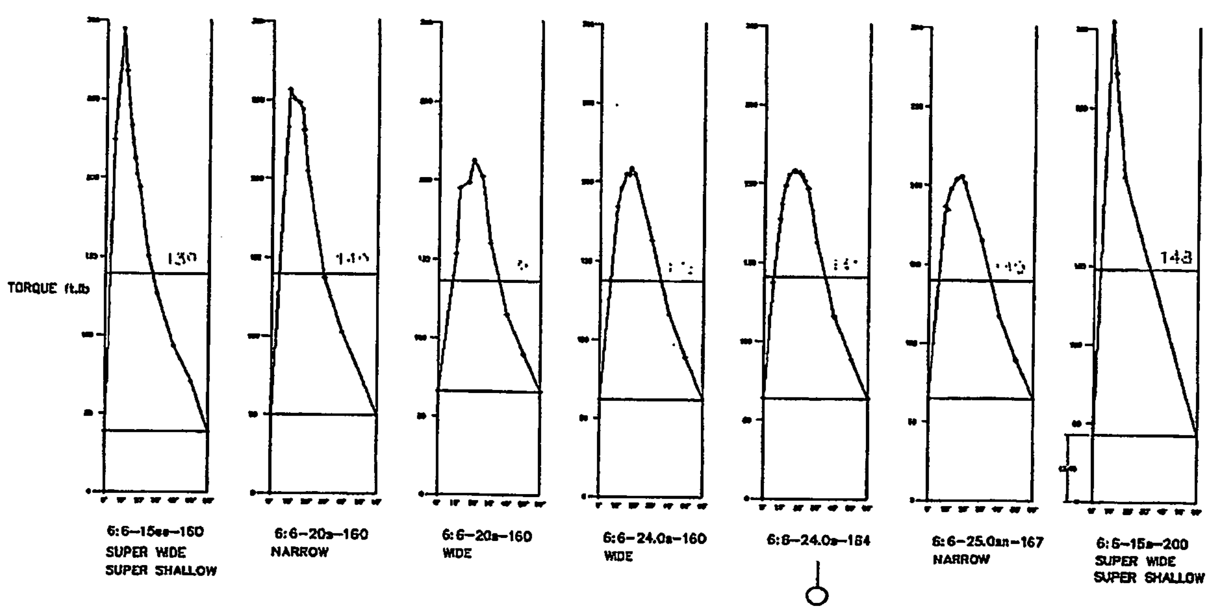
Design of a 1000cc Engine - 1:1 Ratio with 6 teeth.

Proportions of a 1000cc engine.

Profiles of various diameters and tooth depths were developed to compare the graphs of the anticipated torque output for each.

1000cc was the capacity common to all the configurations. The diameter of the male rotor was the basic parameter. The diameters varied from 150mm to 200mm with a number of tooth depths for each diameter. The tooth depth influences the pitch between the rotors and consequently the overall size of the engine.

The following series of graphs show the results of the comparison exercise.



The pressure and torque calculations were based on $P_1.V_1=P_2.V_2$ rather than an indicator diagram as the compression ratio had yet to be determined and this was intended as a comparison exercise only, not an indication of eventual performance.

The comparison procedure indicates that the torque output does not vary much over a wide variety of configurations. It also shows that a wide shallow tooth produces a very high peak early in the cycle. This is not a desirable trait. It is far more advantageous for the torque curve to be as flat and smooth as possible. From this, the obvious choice, as a basis for development, was a profile with a male diameter of 164mm, with a narrow male tooth tip and having as large a tooth depth as possible.

Preliminary. Design

This design would incorporate construction methods like fabrication for the main housing and patternmaking for the endplates. These methods would reduce the cost significantly over steriolithography and investment casting.

Design.

The profiles of the rotors were established during the proportion exercise. This automatically set the length. The next step was to make an initial attempt to establish the timing of the five cycles. Drawing in the first pass at the port sizing and position solidified the overall layout. Cooling, lubrication and air and fuel induction were then added.

It was to be of sandwich construction having the main housing in the centre with the two endplates either side of it and two 12mm aluminium plates on each end of this. There was to be a cast aluminium timing gear cover on the output end and another cast aluminium oil input manifold on the other end. The whole sandwich would be tied together with twelve high tensile tie-bolts.

Detail Design Layout.

Cooling.

The main housing and endplates were to be water cooled with the shafts and rotors, oil cooled. This would require an external water/oil pump unit. This would be a combined centrifugal type unit delivering low-pressure water and oil.

The Main Housing.

The main housing was designed to be fabricated from bright mild steel. The method of manufacture was to roll two 10mm thick tubes allowing a 1.5mm machining allowance. These were to be turned to size on their outer diameter, then cut and welded to form the binocular shape which was to form the basis of the housing. Three 10mm plates were to be laser cut to profile. A series of spacer blocks were to be shaped. Then the whole collection was to be welded to form the main housing assembly. The assembly would then be machined and the inside surfaces case hardened and ground.

The Endplates.

The endplates were designed to be cast in SG400 cast iron.

The Rotors.

The rotors were to be investment cast in SG600 cast iron off wax models produced from steriolithography masters. This was necessary as the detail of the internal coring for the oil cooling was quite complex. To produce the castings from patterns would require complicated core boxes. The rotors were to be bored to be a shrink fit onto the shafts.

The Shafts and Timing Gears.

Because of space restrictions, it was decided to use needle roller bearings running directly on the shaft without an inner race. In order to accomplish this, the shafts were to be manufactured from EN3327 steel, case hardened 0.75 deep to 62 Rockwell C and then fine ground to very fine tolerances.

The timing gears were to be fixed to splines on the output end of the shafts. They were to be helical at 14° to reduce noise and to provide, by shimming, a fine degree of adjustment of the rotor orientation.

The Manifolds.

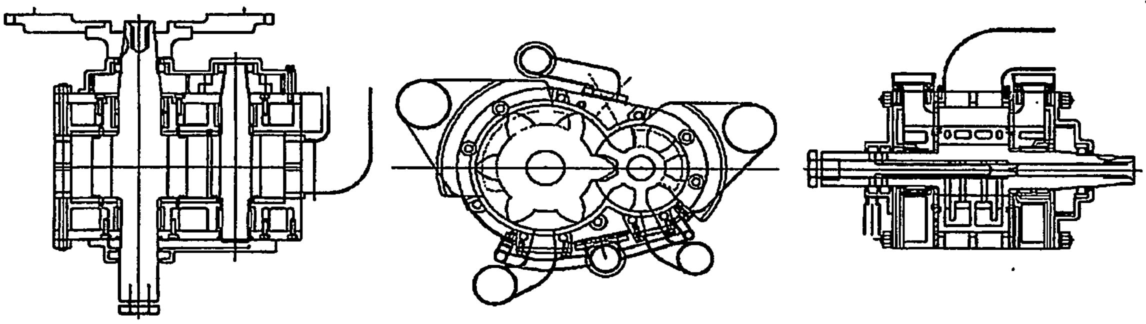
The exhaust manifold would be constructed from stainless steel because the exhaust is almost continuous. This means that the exhaust manifold will run at very high temperature. The manifold plates would be laser cut from 8mm thick stainless steel. The manifold itself would be formed from laser cut 2mm thick stainless steel and TIG welded.

The secondary exhaust (the fifth cycle) would be fabricated from 1mm thick mild steel by the same method.

The inlet system would be constructed in much the same manner except for the throttle assembly which would be turned from aluminium and fitted with a bearing mounted butterfly.

The cooling, inlet and outlet, manifolds would be bent tubing welded to laser cut plates.

The following diagram is a partial view of the 6:6 Ratio - 1000cc Water/Oil Cooled.



6:6 Ratio - 1000cc Oil Cooled.

Oil cooling the main housing simplified the layout. A new design layout was commenced. The water pump and heat exchanger would be eliminated leaving just the oil pump and a larger heat exchanger. The oil inlet to the shafts and rotors would be one-way instead of the two-way water system. The bearings could be lubricated directly from the shafts. The male rotor cooling outlet could be a series of slots in the end faces of the rotor with matching slots in the end plates. The rotor cooling oil would transfer through these slots into the end plates. A partition would partially separate the rotor oil from the main housing flow. The female oil

outlet would be through the far end of the shaft, passing through the timing gear housing and up into the cooling oil outlet manifold. This would mean one common oil outlet manifold.

Design layout. 1000cc 2D CAD Oil Cooled

The rotor profiles and general layout of the water cooled version were retained. The sandwich construction was also preserved. Almost all other aspects underwent a series of refinements and incremental modifications.

Main housing.

The main housing was designed to be fabricated from bright mild steel components welded together to form the complete housing. This would then be machined and the inside surface case hardened with a final fine grind.

End plates.

The end plates were redesigned to be produced by investment casting off steriolithographic master wax models from 3D CAD solid models. This method allows undercuts and wall thickness down to 3mm.

Shafts

The male shaft was simplified by oil cooling and the use of longer, single-row, 50mm diameter, needle roller bearings on both journals. Oil cooling also simplified the oil inlet holes and improved the shaft to rotor oil transfer. The female shaft was redesigned to accept longer needle roller bearings. The timing gear mounting method was changed form a taper fit to involute splines. Involute splines have a gear tooth like shape. The normal practice is to use straight cut splines.

Timing gears.

The timing gears were designed with an 18° helical lead angle. A pair of helical gears are much quieter than spur gears. The more important reason for specifying helical gears, was to introduce a small amount of radial adjustment of the rotors to eliminate rotor-clash. The female gear has two components: a hub mounted on the involute splines, with a ring gear

doweled and screwed to it. The gears are adjusted linearly with respect to each other, by the addition of shims between the hub and ring gear. This system has been used successfully, to adjust rotor clash in roots type superchargers for a very long time.

The amount of radial excursion of the rotors is dependent on the degree of backlash in the gears and the linear movement of the shafts due to end thrust. The end thrust will be taken between the rotor end faces and endplates. The anticipated end clearance is 0.2mm. The variation of rotor orientation caused by backlash in the gears and end thrust variation is absorbed by the spring loaded tip seals in the female rotor.

Bearings

The bearings chosen for the male shaft are 50 mm diameter x 35mm long single row needle roller bearings. These have a nominal dynamic tolerance rating of 4400 kg, which is more than adequate for the task. The nominal rev. limit is 8000 rpm. A radial clearance close to the maximum, 25 microns, is provided. This should allow excursions into the 10,000rpm plus range for short periods of time. The female shaft is fitted with 32mm diameter x 30mm long single row needle roller bearings at each end. These have a dynamic rating of 2950 kg and a nominal rev limit of 12,000 rpm. All four bearings run directly on the shafts without an inner race.

Lubrication and cooling of the bearings.

The bearings are lubricated through holes in the shafts. The shafts have oil feed grooves around the running face. Three of the bearings are cooled by the oil. The bearing at the timing gear end of the female shaft is fed with the hot oil exhausted from the rotor.

Spark plugs.

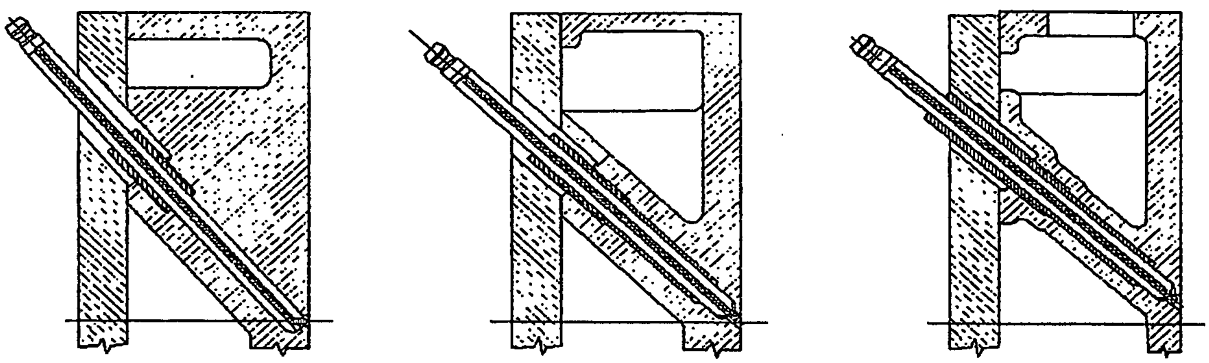
There is a spark plug at each end of the combustion chamber. They fire simultaneously at the appropriate time. The combustion chamber is near the centre of the engine.

Spark plugs designed for a 28cc engine seemed to work quite well, so the starting point for the design of spark plugs to suit this engine was to simply to extend the length of these (Fig. 1).

Fig. 2 shows a conical end on the alumina tube with a straight electrode.

Fig.3 shows the final version with a combined strengthening tube and clamp screw extended, to allow easier access.

The following diagram shows three iterations of the design of the spark plug. It also gives an indication of the transition stages of the endplate.



1 2 3

Induction.

The air is introduced through an air cleaner with a throttle mounted on a plenum chamber. Four ducts connect the plenum chamber to the four inlet galleries. There is a male and female inlet gallery in each endplate leading to the inlet ports. The inlet ports form part of the fifth cycle. The fifth cycle may leave a small residue of burnt gas in the chambers. This should not be a problem as some manufacturers deliberately reintroduce, up to 11% of burnt gas to lower the combustion temperature, as one method to the reduce NO_x emissions. It can reduce the NO_x content up to 60%.

The fuel is introduced by fuel injection early in the compression stage. There are two fuel injectors in the male chamber and two in the female. The two male injectors are set lean. The two female injectors are set rich and are spaced wider apart to impart their charge in the region of the spark plugs. This provides a form of stratified charge.

Cooling.

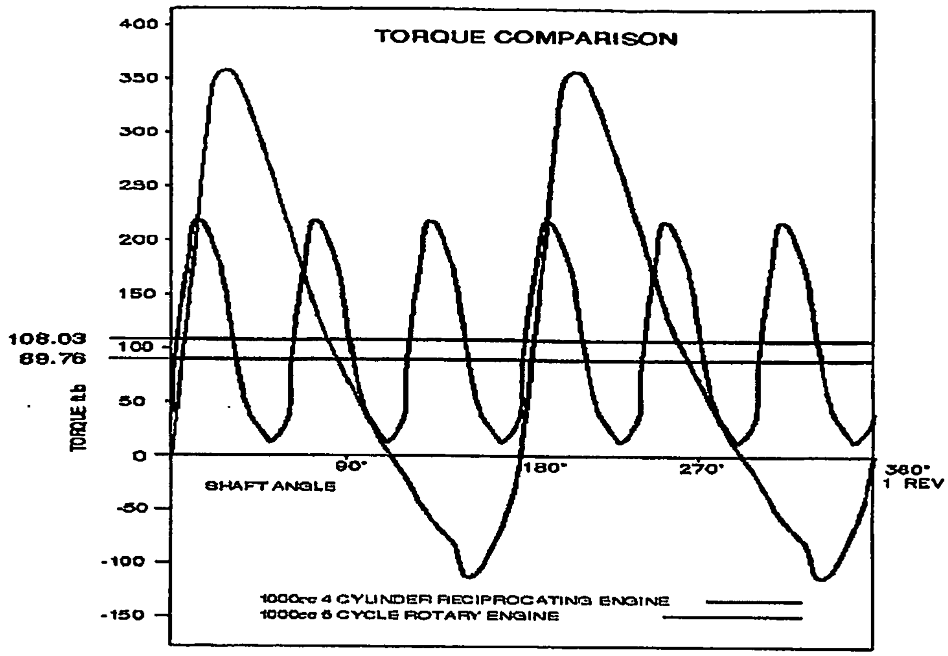
The flow rate of the cooling oil will be varied to maintain the outlet temperature to be about 100°C (212°F). The cooling is introduced first to the hot areas: the exhaust ports and the combustion area. The hot oil is then distributed to some of the cooler areas.

Since combustion takes place six times per revolution, heat absorption by the housing is almost continuous. The rotors cycle through: cold air induction to the heat generated by compression, combustion and expansion. They get a small respite from the heat. The housing has no respite. For this reason, most of the cooling is directed to the housing. It does not matter how hot the housing gets, if the temperature is even over the whole expanse.

The torque of the 5 Cycle 6-1000cc Oil Cooled Rotary Engine.

The torque output of this engine was calculated by first determining the effective area on which the pressure reacts. Then multiplying the effective area by the chamber pressure and the average distance this effective area is from the centre of the shaft. The female rotor shades areas of the male rotor from the pressure while they are engaged. Consequently the effective area is constantly changing. The full area of the male tooth is exposed to the pressure when the rotors disengage. The male tooth is fully exposed to the pressure for 60° of rotation.

The effective area of the male tooth, the chamber volume and the moment to the shaft centre were determined by accurately drawing diagrams of the rotors, at the various rotation angles, using AutoCAD. 2D was adequate to obtain a very accurate result in this instance, as the profiles are linear along the central axes.

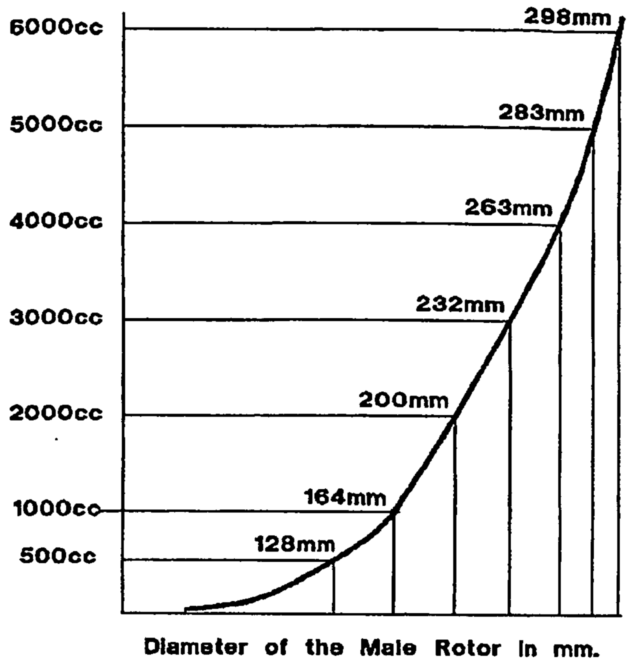


Advantages

1. The power/weight ratio is exceptional. The actual components of the 1000cc prototype weigh 38kg. The materials used in the prototype are all steel or cast iron. The use of light weight materials like aluminium or magnesium alloys would reduce this weight dramatically. It is even possible to use titanium shafts if an application for extremely light weight is justified. This would result in the basic engine weighing about 25kg. The average 2000cc reciprocating engine weighs about 120kg.

2. The power/size ratio is also exceptional. Dimensions of the prototype are 300x300x300 millimetres, including an over large air inlet system. These dimensions will be able to be reduced in subsequent versions. The size advantage is even greater as the capacity of the engine increases. The male rotor of the 1000cc version is 164mm diameter. A 6000cc engine 12,000cc equivalent) would have a male rotor diameter of 298mm.

The following graph gives engine capacity relative to the male rotor diameter.



3. After development, fuel economy should be a feature. The diagram indicates that the 1000cc 5 Cycle Rotary would produce more power, per unit of fuel, than the compared 2000cc reciprocating engine. The results of the comparison show that the 1000cc 5 Cycle Rotary engine might produce 108 ft.lb average torque while the 2000cc reciprocating engine indicated 90 ft.lb average torque. This constitutes a nominal improvement of 20%.

4. Smooth running with minimum vibration constitutes a major feature of this engine.

5. Reduced mechanical noise will be emitted by the exclusion of the whole valve train. Exhaust back-pressure does not affect the power output. This would allow a very quiet exhaust system to be fitted. These attributes should allow the design of a very quiet smooth running engine.

6. A very small component count would reduce production cost considerably. Material usage is dramatically reduced. The energy used to produce the engine should also be very much less than that used to produce an existing engine.

Construction.

Gears, shafts and nuts.

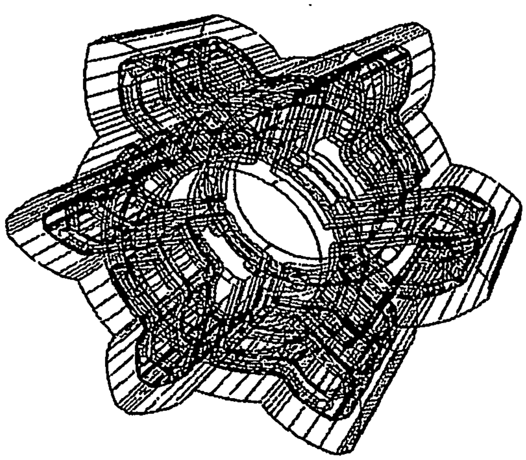
The timing gears, the male and female shafts and the retention nuts were drawn in 2D orthographic drawings.

The shafts were made from EN3327 steel, rough machined, case hardened to 62 Rockwell C, to a depth of 0.75mm and fine ground to very fine tolerances. The case hardening was masked from the threads. Running the needle roller bearings directly on the shafts was to gain the space normally taken by inner races.

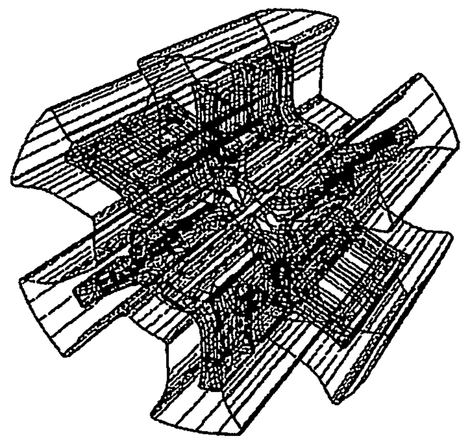
Male and Female Rotors.

The male and female rotors have quite complex internal coring for cooling. The most appropriate method of manufacture was casting directly from steriolithographic wax models. The details of the male and female rotors were taken directly from the 2D CAD design layout, a 1.0mm machining allowance was added and used to develop 3D CAD solid models of the male and female rotor.

Male Rotor 3D CAD Model.

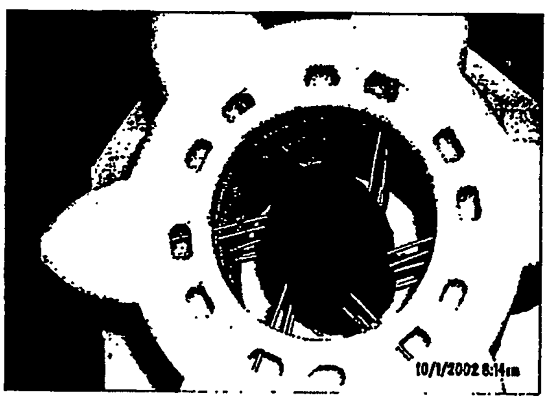


Female Rotor 3D CAD Model.

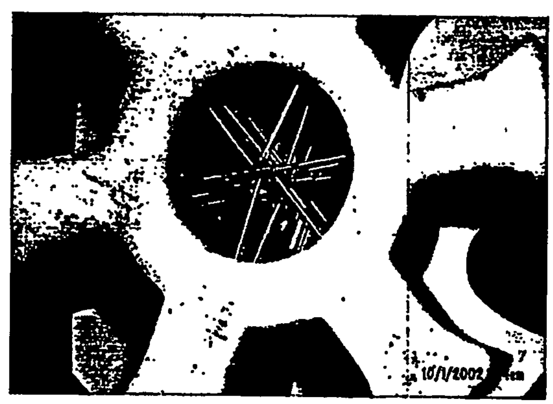


These models were converted to steriolithography files. From these files wax master models were produced, with a shrinkage factor added. The waxes were then prepared for the application of investment.

Male Rotor Wax Master Model.



Female Rotor Wax Master Model.



The wires are to support the extremities of the investment in the cores of the casting. The investments were then fired, which evaporates the wax masters, and used to cast the rotors in SG600 cast iron. The cast iron rotors were then machined. The central bores were ground and mandrels were made to fit with an interference fit to the rotors. The mandrels were necessary in order to mount the rotors on a dummy shaft to 4 axis NC mill the profiles, to

face them to length and turn the circular sealing grooves in both. The NC milling profiles were taken directly from the original 2D CAD design layout, and transferred to the milling operation as '.igs' extension files.

The tip seal grooves were milled using a slit cutter and the crucial end seal grooves were added using the Electrode Discharge Machining (EDM) process, and the individual end seals were hand lapped to fit.

The male and female rotors were designed to be a shrink fit on the shafts. The degree of shrink fit was 0.025mm per 25mm diameter.

Main housing.

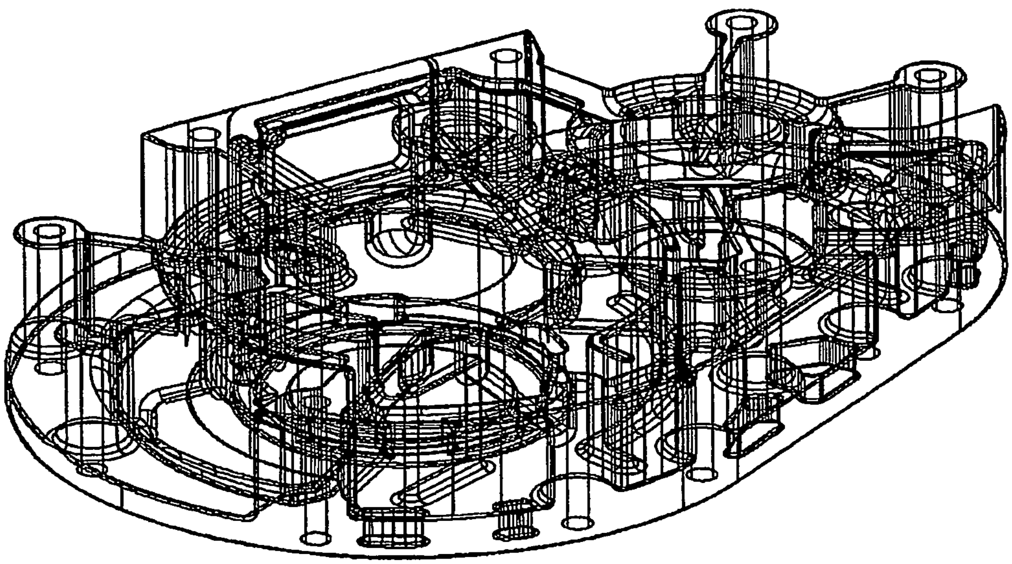
The main housing was fabricated from bright mild steel. Two rings were rolled from 10mm thick plate. The diameters of the rings were appropriate to allow 2mm diameter machining allowance on the outside and 6mm diameter on the bores. The rings were machined on the outside, then cut and scarf welded to form the binocular shape of the housing. Two profiles were laser cut from 10mm plate and one form 8mm. These were welded to the outside of the binocular shape to form flanges. Sixteen blocks and ten tubular spacers were machined and inserted between the plates. The assembly was then welded to form the basis of the main housing. The excess weld material in the ports and around the posts was removed by hand filing. The bores were rough machined and the end surfaces were faced. The housing was then stress relieved. The bores and the end faces were machined again to within 0.25mm of the final size. The four fuel injector ports were milled. The bores were case hardened to 60 Rockwell C and finally ground to a fine finish and close tolerance. The tip seal landing zone ramps were stoned by hand. The bolt holes were drilled and five of them were reamed to accept specially ground tie bolts to act as dowles. The cooling oil holes were added in the end flanges and three oil inlet manifold plates were welded in place to form the outer cladding. The final act was to drill and tap the holes to accept mounting screws for the male and female primary exhausts and fifth cycle ducting.

Endplates.

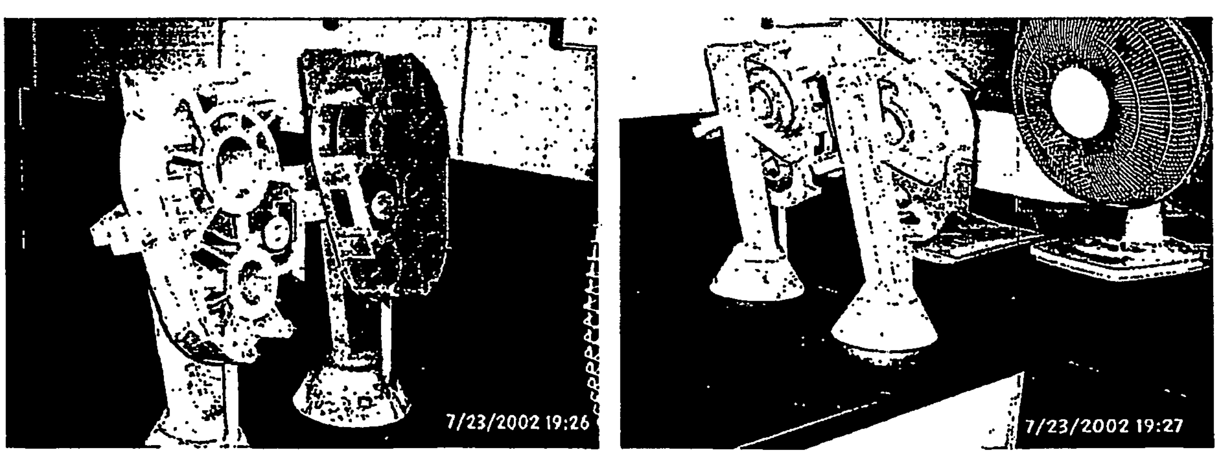
The endplates were the mirror image of each other and after the conversion from the steriolithography file to the file format used by the steriolithography machine the shrinkage

factor was added and support flanges inserted to enable undercuts. This process allows the design of the parts to closely approximate that of high quantity production techniques. It allowed the addition of wider sealing flanges on the outer edges of the ribs, a general wall thickness reduction and elimination of excess material around the spark plug.

Endplate 3D CAD Solid Model.



The Endplate Waxes with one layer of investment.



The photographs show the wax masters being prepared in a 'tree'. The investment coating was built up to be approximately 10mm thick prior to firing.

Cover plates.

The Endplate oil cooling cavities are formed by two 12mm thick aluminium cover plates. There is one on each end and are extended downwards to become the engine mounts. The cover plates were milled to carry the shaft oil seals, and drilled and tapped to accept the oil inlet manifold, on the left hand end, and the timing gear housing on the output end. The oil pump is mounted to the left hand cover plate. The cover plates are the bread of the sandwich construction.

Timing Gear Housing and Oil Inlet Manifold.

The timing gear housing and the oil inlet manifold are both fabricated using 6mm thick laser cut mild steel plates. The housing is constructed from 2mm thick sheet brazed to the 6mm plate. The oil inlet manifold is a little more complex. It has four inlet ports and is made up of four separate brazed assemblies screwed together to form the manifold. The two extra inlet ports cooled the two small cavities on the far sides of the exhaust ports.

Air Inlet System.

The air inlet system is composed of an air filter, a throttle assembly mounted on a plenum chamber. Four ducts connect the plenum chamber to the inlet ports in the endplates. The plenum chamber is clamped to the four inlet ducts with four toggle action clips. This allows quick access to the fuel injectors.

The whole assembly is formed from 1mm sheet welded and brazed to 3mm thick mounting plates.

Exhaust Manifolds.

The primary exhaust manifolds are constructed from 2mm stainless steel sheet formed and TIG welded. These assemblies are TIG welded to 8mm thick laser cut manifold plates.

The secondary exhaust manifolds, associated with the 5th cycle, are constructed from 1mm mild steel sheet formed and welded. These are, in turn, welded to 3mm manifold plates.

Ignition system.

A programmable ignition computer was added to the CDI unit and a tachometer. This provides the automatic advance and retard facility to the ignition timing.
The spark plugs were constructed from 6mm outside diameter x 2mm bore alumina ceramic tubing. A 2mm diameter steel electrode is sealed in position with alumina ceramic adhesive. A threaded external support tube is secured around the tube with the same adhesive. Conical ends were formed on the tubes to dress the grinding wheel.

Fuel Introduction System.

The fuel is introduced through four fuel injectors mounted in extension tubes fitted to the main housing. The extension tubes allow the four pintles to be as closely spaced as required. The internal bores of the extension tubes are tapered to help prevent 'wetting' of the atomised fuel.

Throughout the specification, unless the context requires otherwise, the word "comprise", and variations such as "comprises" or "comprising", will be understood to imply the inclusion of a stated step or integer or group of steps or integers but not the exclusion of any other step or integer or group of steps or integers.

The reference to any prior art in this specification is not, and should not be taken as, an acknowledgement or any form of suggestion that that prior art forms part of the common general knowledge in Australia.

The embodiments have been described by way of example only and modifications are possible within the scope of the invention.

DATED this 8th day of October, 2002

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